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(54) **TWO-SPEED FULL-TIME TRANSFER CASE WITH INTEGRATED PLANETARY GEARSET AND SYNCHRONIZED RANGE SHIFT**

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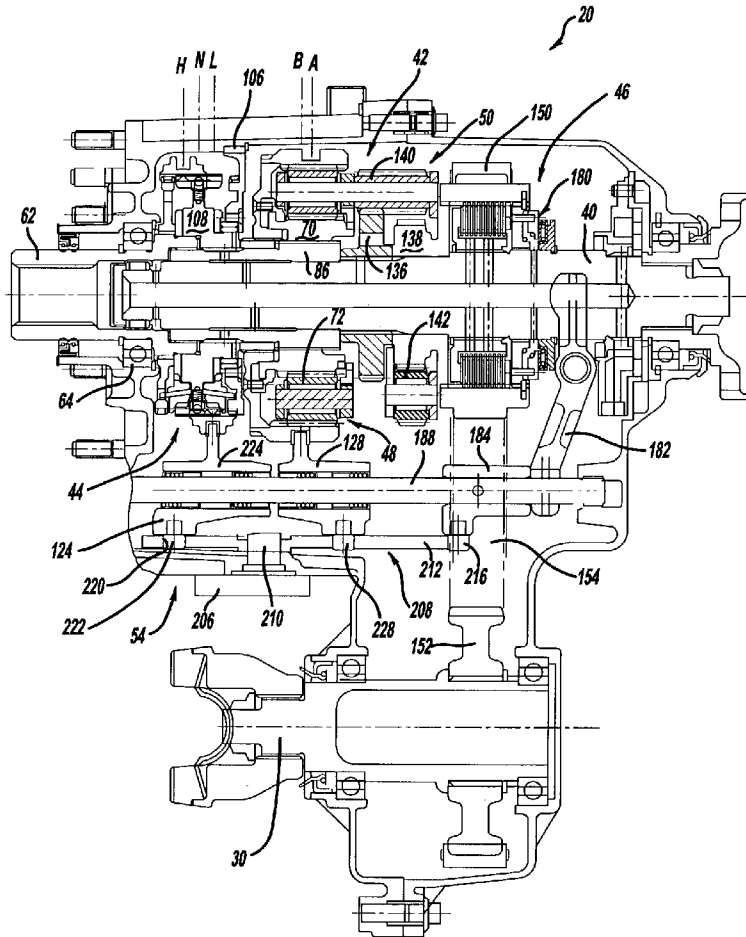
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ABSTRACT

(57) A full-time transfer case is equipped with an integrated planetary gearset assembly and a synchronized range shift mechanism to provide high-range and low-range drive connections. The integrated planetary gearset assembly includes a first gearset which acts as a two-speed reduction unit and a second gearset which acts as an interaxle differential. The synchronized range shift mechanism is arranged to concurrently move two components of the first gearset and is synchronized to permit on-the-move range shifts.

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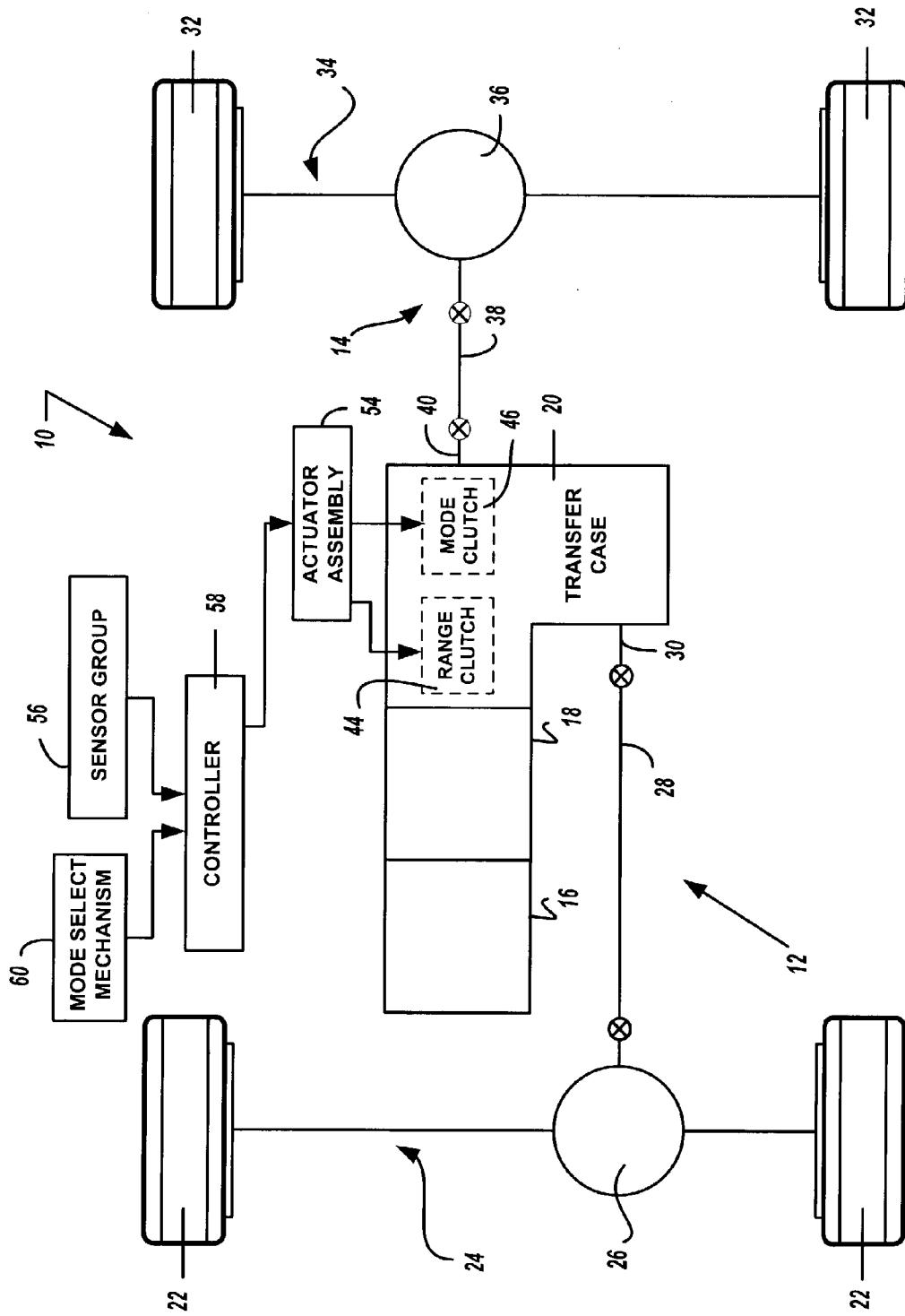


Figure - 1

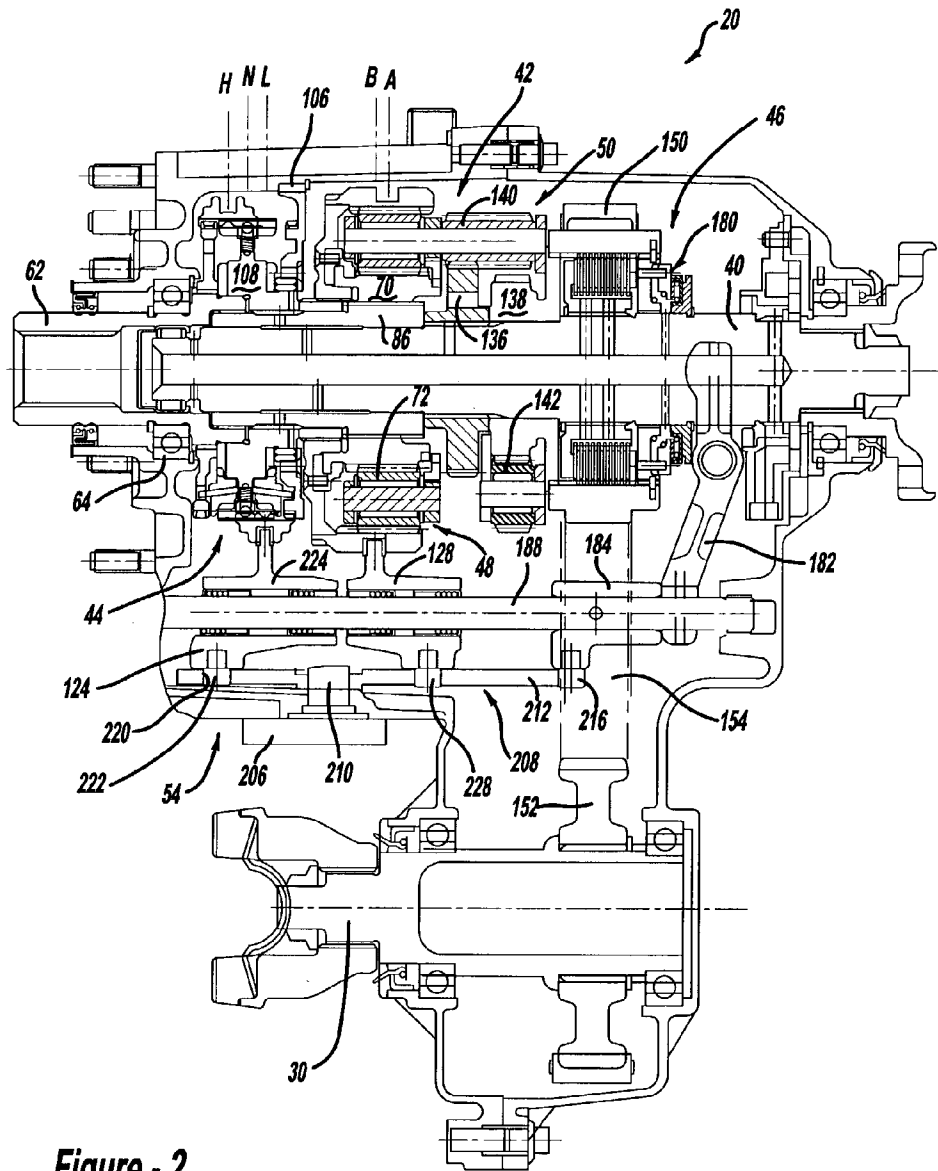


Figure - 2

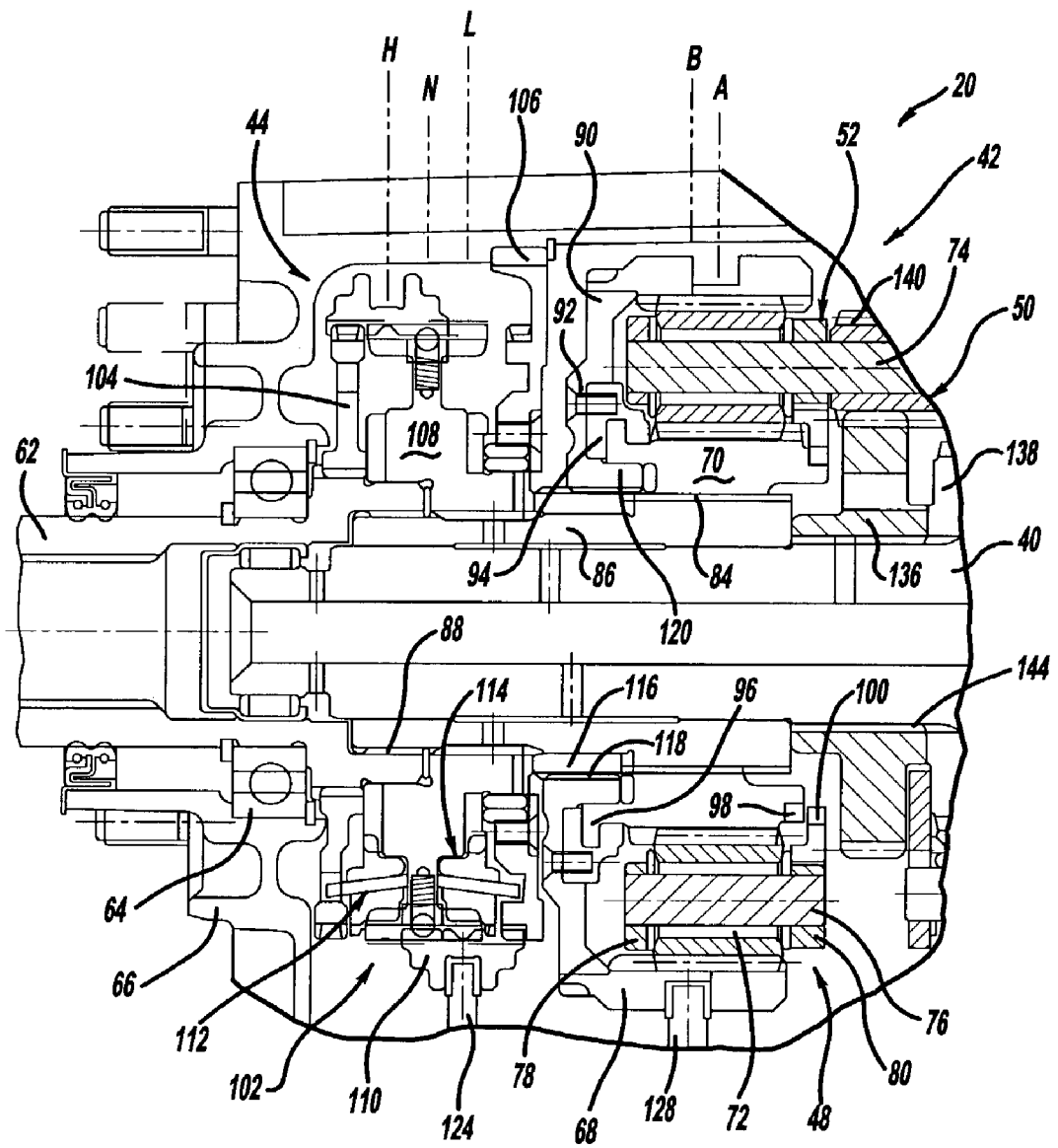


Figure - 3

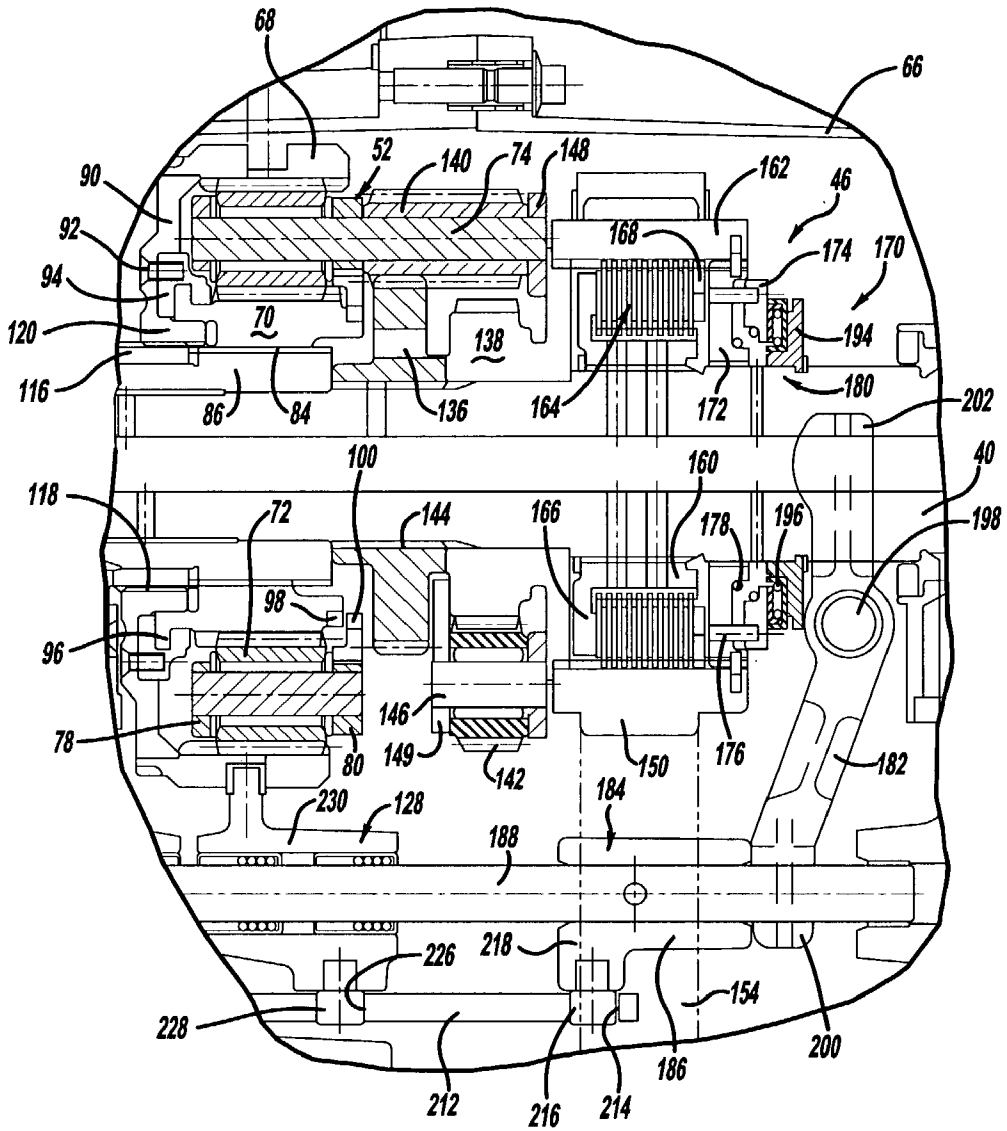


Figure - 4

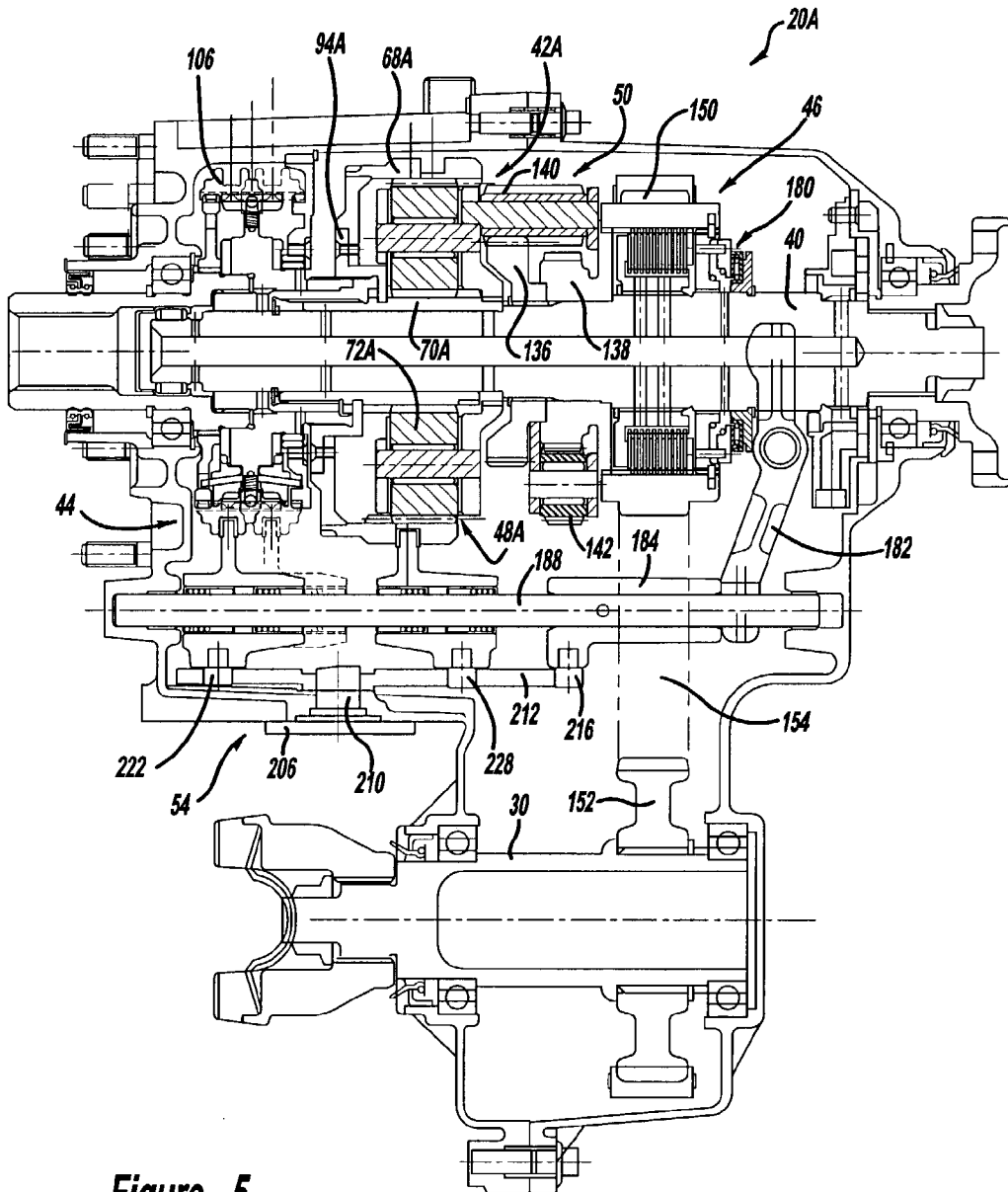


Figure - 5

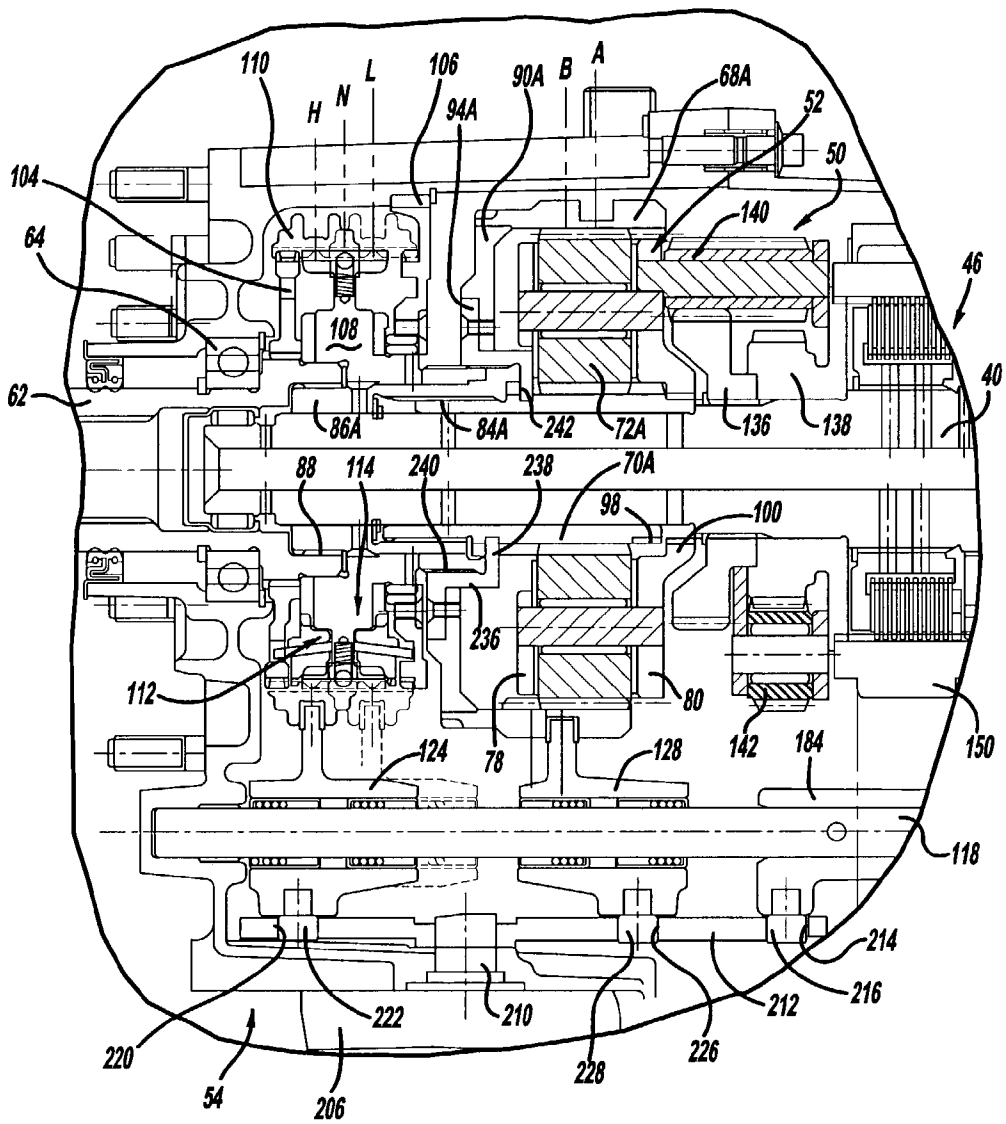


Figure - 6

TWO-SPEED FULL-TIME TRANSFER CASE WITH INTEGRATED PLANETARY GEARSET AND SYNCHRONIZED RANGE SHIFT

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application is a divisional of U.S. patent application Ser. No. 09/981,164 filed on Oct. 17, 2001.

FIELD OF THE INVENTION

[0002] The present invention relates generally to a power transfer system for controlling the distribution of drive torque between the front and rear drivelines of a four-wheel drive vehicle. More particularly, the present relates to a full-time transfer case having a two-speed gear reduction unit and an interaxle differential integrated into a planetary gear assembly, a synchronized range shift mechanism for establishing high-range and low-range drive modes, and a biasing clutch for controlling interaxle slip between the front and rear drivelines.

BACKGROUND OF THE INVENTION

[0003] Due to growing consumer demand for four-wheel drive vehicles, a plethora of different power transfer systems are currently utilized for directing power (i.e., drive torque) to all four wheels of the vehicle. For example, in many "part-time" power transfer systems, a transfer case is installed between the front and rear drivelines and is normally operable in a two-wheel drive mode for delivering drive torque to the driven wheels. However, the four-wheel drive mode is desired, a mode shift mechanism is selectively actuated by the vehicle operator for directly coupling the non-driven wheels to the driven wheels for establishing a part-time or locked four-wheel drive mode. One example of a part-time transfer case is disclosed in commonly-owned U.S. Pat. No. 4,770,280.

[0004] It is also known to use "on-demand" power transfer systems for automatically directing power to the non-driven wheels, without any input or action on the part of the vehicle operator, when traction is lost at the driven wheels. Typically, the on-demand feature is incorporated into the transfer case by replacing the mode shift mechanism with a clutch assembly that is interactively associated with an electronic control system and a sensor arrangement. During normal road conditions, the clutch assembly is maintained in a non-actuated condition such that drive torque is only delivered to the driven wheels. However, when the sensors detect a low traction condition at the driven wheels, the clutch assembly is automatically actuated to deliver drive torque to the non-driven wheels. The amount of drive torque transferred through the clutch assembly can be varied as a function of specific vehicle dynamics detected by the sensor arrangement. One example of an on-demand power transfer system is disclosed in commonly-owned U.S. Pat. No. 5,323,871.

[0005] As yet a further alternative, some vehicles are equipped with a full-time power transfer system having a transfer case equipped with a center differential that functions to permit interaxle speed differentiation while transferring drive torque to both the front and rear drivelines. To minimize loss of traction due to wheel slip, many full-time transfer cases are also equipped with a clutch assembly for

limiting speed differentiation and biasing the torque transferred across the center differential. For example, full-time transfer cases are disclosed in commonly-owned U.S. Pat. Nos. 5,697,861 and 5,702,321.

[0006] A significant number of the transfer cases discussed above are equipped with a gear reduction unit and a range shift mechanism operable for permitting the vehicle operator to choose between high-range and low-range drive modes. In many instances, the vehicle must be stopped before the transfer case can be shifted between its high-range and low-range drive modes. Unfortunately, the need to stop the vehicle prior to shifting between the high-range and low-range drive modes is inconvenient, particularly upon encountering road conditions or surface terrains where continuation of the vehicle's rolling momentum would assist in overcoming the conditions encountered. To alleviate this inconvenience, some two-speed transfer cases are equipped with a synchronized range shift mechanism for permitting "on-the-move" shifting between the high and low ranges.

[0007] In an effort to minimize the overall size of full-time two-speed transfer cases, it has been proposed to incorporate the gear reduction unit and the interaxle differential into a common planetary gear assembly. For example, commonly-owned U.S. Pat. No. 5,902,205 discloses a full-time two-speed transfer case equipped with an integrated planetary gearset which is operable for establishing full-time high-range and low-range four-wheel drive modes through on-the-move shifting of a synchronized range shift mechanism. While such an arrangement provides a compact construction, there is a continuing need to develop alternatives which meet modern requirements for low noise and weight while advancing the state of the four-wheel drive art.

SUMMARY OF THE INVENTION

[0008] It is therefore an object of the present invention to provide a transfer case for a full-time four-wheel drive vehicle having a planetary gear assembly which integrates a gear reduction unit and an interaxle differential into a common arrangement.

[0009] As an additional object of the present invention, the full-time two-speed transfer case includes a synchronized range shift mechanism which can be selectively actuated for establishing a full-time four-wheel high-range drive mode, a neutral mode, and a full-time four-wheel low-range drive mode.

[0010] According to another object of the present invention, the transfer case includes a biasing clutch which is operably associated with the outputs of the planetary gear assembly for limiting speed differentiation and regulating the drive torque distribution therebetween in response to the occurrence of slip between the front and rear output shafts of the transfer case. To this end, a control system is provided which includes sensors for detecting and generating sensor signals indicative of various dynamic and operational characteristics of the vehicle, and a controller for controlling actuation of the biasing clutch in response to the sensor signals. Upon the occurrence of traction loss, the clutch is automatically actuated for limiting interaxle slip while transferring increased drive torque to the non-slipping driveline.

[0011] According a preferred embodiment of the present invention, the planetary gear assembly is operably installed

between and input shaft and front and rear output shafts of the transfer case and is constructed in a compact arrangement. The planetary gear assembly includes a first planetary gearset and a second planetary gearset which are interconnected by a common carrier assembly. The first planetary gearset is operably installed between the input shaft and the second planetary gearset for driving the carrier assembly at either of a direct speed ratio (i.e., high-range) or a reduced speed ratio (i.e., low-range) relative to the input shaft. The common carrier assembly acts as the input to the second planetary gearset which has first and second outputs respectively connected to the rear and front output shafts of the transfer case. Thus, the second planetary gearset functions as an interaxle differential for permitting speed differentiation and distributing drive torque between the front and rear output shafts of the transfer case.

[0012] Additional objects come with features and advantages of the present invention will become apparent from studying the following detailed description and appended claims when taken in conjunction with accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013] FIG. 1 is a schematic representation of a four-wheel drive motor vehicle equipped with a full-time power transfer system according to the present invention;

[0014] FIG. 2 is a sectional view of a full-time two-speed transfer case constructed according to a preferred embodiment of the present invention;

[0015] FIG. 3 is an enlarged partial view taken from FIG. 2 showing components of the synchronized range shift mechanism in greater detail;

[0016] FIG. 4 is an enlarged partial view taken from FIG. 2 showing the components of the integrated planetary gear assembly in greater detail;

[0017] FIG. 5 is a sectional view of a full-time two-speed transfer case constructed according to an alternative preferred embodiment of the present invention;

[0018] FIG. 6 is an enlarged partial view of FIG. 5 showing the synchronized range shift mechanism and integrated planetary gear assembly in greater detail; and

[0019] FIG. 7 is a partial sectional view showing another alternative preferred embodiment of a synchronized range shift mechanism and integrated planetary gear assembly for use in a full-time two-speed transfer case of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0020] Referring now to the drawings, a drivetrain for a four-wheel drive vehicle is schematically shown interactively associated with a power transfer system 10 of the present invention. The motor vehicle drivetrain includes a front driveline 12 and a rear driveline 14 both driveable from a source of power, such as an engine 16, through a transmission 18 which may be of the manual or automatic type. In the particular embodiment shown, the drivetrain includes a transfer case 20 for transmitting drive torque from engine 16 and transmission 18 to front driveline 12 and rear driveline 14. Front driveline 12 includes a pair of front

wheels 22 connected at opposite ends of a front axle assembly 24 having a front differential 26 coupled to one end of a front drive shaft 28, the opposite end of which is coupled to a front output shaft 30 of transfer case 20. Similarly, rear driveline 14 includes a pair of rear wheels 32 connected at opposite ends of a rear axle assembly 34 having a rear differential 36 coupled to one end of a rear drive shaft 38, the opposite end of which is interconnected to a rear output shaft 40 of transfer case 20.

[0021] As will be detailed hereinafter with greater specificity, transfer case 20 is equipped with a planetary gear assembly 42, a range clutch assembly 44, and a mode clutch assembly 46. Planetary gear assembly 42 includes a first planetary gearset 48 and a second planetary gearset 50 interconnected through a common carrier assembly 52. Range clutch assembly 44 is operable to couple components of first planetary gearset 48 for driving carrier assembly 52 at either of a first (high-range) speed ratios or a second (low-range) speed ratio. Second planetary gearset 50 of planetary gear assembly 42 functions as an interaxle differential having carrier assembly 52 as its input with other components thereof operably coupled to front output shaft 30 and rear output shaft 40 of transfer case 20. Additionally, mode clutch assembly 48 is operable to control the magnitude of speed differentiation and torque biasing between rear output shaft 40 and rear output shaft 30. Power transfer system 10 further includes a power-operated actuator assembly 54 for controlling actuation of range clutch assembly 44 and mode clutch assembly 46, a sensor group 56 for sensing dynamic and operational characteristics of the motor vehicle and generating sensor input signals indicative thereof, and a controller 58 for generating control signals in response to the sensor input signals. Power transfer system 10 also includes a mode select mechanism 60 for permitting the vehicle operator to select one of the available drive modes. In particular, controller 58 functions to control actuation of power-operated actuator assembly 54 in response to the mode signal sent to controller 58 from mode select mechanism 60 that is indicative to the particular mode selected.

[0022] With particular reference now to FIGS. 2 through 4 of the drawings, transfer case 20 is shown to include an input shaft 62 rotatably supported by a bearing assembly 64 from a housing 66. Input shaft 62 is adapted for connection to an output shaft of transmission 18. Likewise, front output shaft 30 and rear output shaft 40 are also rotatably supported by housing 66. As noted, planetary gear assembly 42 includes a first gearset 48 interconnected to second gearset 50 via a common carrier assembly 52. First gearset 48 includes a ring gear 68, a first sun gear 70, and a set of first planet gears 72 each meshed with ring gear 68 and first sun gear 70. Planet gears 72 are rotatably supported on long pins 74 and short pins 76, each of which extends between first and second carrier rings 78 and 80, respectively, of carrier assembly 52. First sun gear 70 is shown fixed via a splined connection 84 for rotation with and axial sliding movement on a quill shaft 86 journaled on rear output shaft 40 and which is fixed via a splined connection 88 for rotation with input shaft 62. As such, first sun gear 70 is always driven by input shaft 62.

[0023] Ring gear 68 is shown to include a radial plate segment 90 that is fixed via bolts 92 to a coupling ring 94. Coupling ring 94 includes a circumferential groove within which a radial lug segment 96 of first sun gear 70 is retained.

Coupling ring 94 permits first sun gear 70 to rotate freely relative to ring gear 68 while also enabling concurrent axial movement of ring gear 68 and sun gear 70 relative to carrier assembly 52 between a first position (denoted by position line "A") and a second position (denoted by position line "B"). When ring gear 68 and first sun gear 70 are located in the A position, external clutch teeth 98 on first sun gear 70 are engaged with internal clutch teeth 100 on second carrier ring 80. As such, first sun gear 70 couples carrier assembly 52 for common rotation with input shaft 62. In contrast, sun gear clutch teeth 98 are released from engagement with clutch teeth 100 on second carrier ring 80 when ring gear 68 and first sun gear 70 are located in the B position, thereby allowing carrier assembly 52 to rotate relative to input shaft 62.

[0024] Range clutch assembly 44 is shown to include a range shift mechanism 102 having a first clutch plate 104 fixed (i.e., splined) to input shaft 62, a second clutch plate 106 fixed to housing 66, a clutch hub 108 journalled on portions of input shaft 62 and quill shaft 86, and a dog-type range sleeve 110 that is splined to clutch hub 108 for rotation therewith and axial movement thereon between three distinct range positions. A non-synchronized version of range shift mechanism 102 is shown in the upper-half of FIG. 2. Preferably, however, range shift mechanism 102 is synchronized to permit "on-the-move" shifting. Thus, a synchronized version of range shift mechanism 102 is shown in the lower-half of FIG. 2. The synchronized version of range shift mechanism 102 includes a first synchronizer 112 disposed between clutch hub 108 and first clutch plate 104 and a second synchronizer 114 disposed between clutch hub 108 and second clutch plate 106. Clutch hub 108 is shown to include an axially extending shaft segment 116 that is journalled on quill shaft 86. Ring gear 68 is coupled to clutch hub 108 via a splined connection 118 provided between shaft segment 116 of clutch hub 108 and an axially-extending hub 120 formed on plate segment 90 of ring gear 68. Splined connection 118 permits axial movement of ring gear 68 between its A and B positions while maintaining a mechanical coupling with clutch hub 108.

[0025] As noted, range sleeve 110 is moveable between three distinct range positions shown to include a high-range ("H") position, a low-range ("L") position, and a neutral ("N") position. To this end, a first range fork 124 is provided which moves under the control of actuator assembly 54 to control axial movement of range sleeve 110 between its three range positions. First range fork 124 has a fork segment retained in a groove formed in range sleeve 110. Likewise, a second range fork 128 is provided which also moves under the control of actuator assembly 54 to control axial movement of ring gear 68 and first sun gear 70 between the two distinct positions. As seen, second range fork 128 has a fork segment retained in a groove formed in the outer periphery of ring gear 68. As will be detailed, actuator assembly 54 includes a drive mechanism which functions to coordinate movement of range forks 124 and 128 so as to coordinate axial movement of range sleeve 110 and ring gear 68 to establish different drive connections between input shaft 62 and carrier assembly 52.

[0026] A first or high-range drive connection is established between input shaft 62 and carrier assembly 52 when range sleeve 110 is in its H position and ring gear 64 is in its A position, as shown in the upper half of FIG. 2. With range

sleeve 110 in its H position, its internal clutch teeth and engaged with external clutch teeth on first clutch plate 104 such that clutch hub 108 and ring gear 68 are coupled for common rotation with input shaft 62. Concurrently, positioning of ring gear 68 and first sun gear 70 in the A position causes external clutch teeth 98 on first sun gear 70 to engage internal clutch teeth 100 on second carrier ring 80. Thus, ring gear 68, first sun gear 70 and carrier assembly 52 are all commonly driven by input shaft 62. This establishes a direct speed ratio drive connection between input shaft 62 and carrier assembly 52 such that first planet gears 72 are unloaded during operation of transfer case 20 in the high-range modes. This is a significant advantage over conventional two-speed planetary units since it eliminates concerns about fretting corrosion of the teeth on first planet gears 72 and permits use of quieter and smaller helical gears instead of conventional spur gears.

[0027] A second or low-range drive connection is established between input shaft 62 and carrier assembly 52 when range sleeve 110 is in its L position and ring gear 68 is in its B position, as is shown in the lower-half of FIG. 2. With range sleeve 110 in its L position, its internal clutch teeth are engaged with external clutch teeth formed on second clutch plate 106 such that clutch hub 108 and ring gear 68 are braked against housing 66 to prevent rotation. In addition, movement of ring gear 68 and first sun gear 70 to the B position causes first sun gear 70 to slide axially to a position whereat its clutch teeth 98 are disengaged from clutch teeth 100 on second carrier ring 80. As such, driven rotation of input shaft 62 drives first sun gear 70 via quill shaft 86 such that carrier assembly 52 is rotatively driven at a reduced speed due to ring gear 68 being braked. Finally, a neutral (non-driven) mode is established when range sleeve 110 is in its N position and ring gear 68 is in its A position. With range sleeve 110 in its N position, its internal clutch teeth are disengaged from the external clutch teeth on first and second clutch plates 104 and 106. In this mode, no drive torque is delivered from input shaft 62 to carrier assembly 52.

[0028] With continued reference to FIGS. 2 through 4, second gearset 50 is shown as a dual-planetary arrangement which functions as an interaxle differential to permit speed differentiation and distribute drive torque between front output shaft 30 and rear output shaft 40. Carrier assembly 52, when driven at either of the high-range and low-range speed ratios, acts as the input to second gearset 50 which has its outputs coupled to output shafts 30 and 40. In particular, second gearset 50 includes a second sun gear 136, a third sun gear 138, a set of full-length second planet gears 140, and a set of half-length third planet gears 142. Second sun gear 136 is shown to be fixed via a splined connection 144 to rear output shaft 40 while third sun gear 138 is journally supported thereon. Second planet gears 140 are rotatably supported on long pins 74 while third planet gears 142 are rotatably supported on short pins 146. Long pins 74 are shown to extend between second carrier ring 80 and a third carrier ring 148 while short pins 146 are shown to extend between third carrier ring 148 and a fourth carrier ring 149. Second planet gears 140 are meshed with second sun gear 136 while third planet gears 142 are meshed with third sun gear 138. In addition, second and third planet gears 140 and 142 are circumferentially arranged in meshed pairs. According to the particular construction shown, second sun gear 136 acts as a first output of second gearset 50 while third sun gear 138 acts as the second output thereof. Third sun gear

138 is fixed to a drive sprocket **150** that is operable for transferring drive torque to a driven sprocket **152** fixed to front output shaft **30**. A power chain **154** is shown to interconnect driven sprocket **152** to drive sprocket **150**.

[0029] As best seen from FIG. 4, mode clutch assembly **46** is arranged to control speed differentiation and torque biasing between front output shaft **30** and rear output shaft **40**. Mode clutch assembly **46** is a multi-plate friction clutch which includes a clutch hub **160** that is fixed to rear output shaft **40**, a clutch drum **162** integrally formed with drive sprocket **150**, and a clutch pack **164** disposed therebetween. Clutch pack **164** includes a set of inner friction plates splined to clutch hub **160** and which are interleaved with a set of outer friction plates splined to drum **162**. Clutch pack **164** is located between a reaction plate **166** formed integrally with clutch hub **160** and an apply plate **168** that is splined to drum **162**. As will be detailed, movement of apply plate **168** relative to clutch pack **164** functions to vary the compressive engagement force exerted thereon for adaptively regulating speed differentiation and torque biasing between front output shaft **30** and rear output shaft **40**.

[0030] Mode clutch assembly **46** further includes a clutch actuation mechanism **170** for moving apply plate **168** under the control of actuation assembly **54**. Mechanism **170** includes a locator plate **172** that is splined for rotation with clutch drum **162**, a pressure plate **174**, and a set of thrust pins **176** having one end fixed to pressure plate **174** and which extend through apertures in locator plate **172**. The second end of thrust pins **176** are adapted to engage apply plate **168**. A return spring **178** urges pressure plate **174** in a direction away from locator plate **172** for normally retracting thrust pins **176** from engagement with apply plate **168**. However, axial movement of pressure plate **174** between a fully-retracted position and a fully-extended position causes thrust pins **176** to exert a clutch engagement force on apply plate **168** that varies between predetermined minimum and maximum values.

[0031] To provide means for moving pressure plate **174** between its fully retracted and fully extended positions, clutch actuation mechanism **170** includes a thrust assembly **180**, a lever arm **182**, and a mode fork **184**. Mode fork **184** has a tubular segment **186** fixed to a shift rail **188**, the opposite ends of which are supported in sockets formed in housing **66**. Thrust assembly **180** includes a thrust ring **194** and a thrust bearing assembly **196** that is disposed between thrust ring **194** and pressure plate **174**. In addition, lever arm **182** is mounted to a pivot post **198** for pivotal movement relative to thrust assembly **180**. Lever arm **182** includes a first end portion **200** that is journalled on shift rail **188** and which engages one end of mode fork **184**. The opposite end portion **202** of lever arm **182** is a C-shaped bifurcated section partially surrounding rear output shaft **40** and which engages thrust ring **194**. In operation, axial movement of mode fork **184** causes corresponding pivotal movement of lever arm **182** which, in turn, controls movement of thrust assembly **180** and pressure plate **174**.

[0032] Preferably, actuator assembly **54** includes a rotary actuator, such as an electric gearmotor **206**, which is operable for generating an output torque, the value of which varies as a function of the magnitude of the electrical control signal applied thereto by controller **60**. To provide means for selectively controlling the magnitude of the clutch engage-

ment force exerted on clutch pack **164** and coordinate movement of range forks **124** and **128**, actuator assembly **54** further includes a drive mechanism **208**. Drive mechanism **208** is interconnected to a rotary output member **210** of gearmotor **206** for changing its output torque into an axially-directed forces that are used for controlling axial movement of range forks **124**, **128** and mode fork **184**. According to a preferred construction, drive mechanism **208** includes a sector plate **212** that is rotatably driven through a range of angular motion by output member **210** of gearmotor **206**.

[0033] To generate axial movement of mode fork **184**, sector plate **212** includes a mode slot **214** within which a mode pin **216** is retained. Mode pin **216** is fixed to a flange section **218** of mode fork **184**. The contour of mode slot **214** is configured to cause the desired direction and amount of axial movement of mode fork **184** in response to rotation of sector plate **212** for generating the desired clutch engagement force exerted by actuation mechanism **170** on clutch pack **164**. To control axial movement of range sleeve **110**, sector plate **212** also has a first range slot **220** within which a first range pin **222** extends. First range pin **222** is fixed to a tubular segment **224** of first range fork **124** which is shown supported for sliding movement on shift rail **188**. The contour of first range slot **220** is configured to cause controlled axial movement of range sleeve **110** in response to controlled rotation of sector plate **212**. In a similar fashion, sector plate **212** includes a second range slot **226** within which a second range pin **228** extends. Second range pin **228** is fixed to a tubular segment **230** of second range fork **128** which is likewise supported for sliding movement on shift rail **188**. Again, the contour of second range slot **226** is configured to cause control axial movement of ring gear **68** and first sun gear **70** in response to controlled rotation of sector plate **212**.

[0034] According to a preferred embodiment of the present invention, sector plate **212** may be rotated to any one of five distinct sector positions to establish a corresponding number of drive modes. These modes may include a locked four-wheel high-range drive mode, a full-time four-wheel high-range drive mode, a neutral mode, a full-time four-wheel low-range drive mode, and a locked four-wheel low-range drive mode. The particular four-wheel drive mode selected is established by the position of mode pin **216** in mode slot **214**, the position of first range pin **222** in first range slot **220**, and the position of second range pin **228** and second range slot **226**. In operation, the vehicle operator selects a desired four-wheel drive mode via actuation of mode select mechanism **60** which, in turn, sends a mode signal to controller **58** that is indicative of the selection. Thereafter, controller **58** generates an electric control signal that is applied to gearmotor **206** for controlling the rotated position of sector plate **212**. More particularly, upon selection of the locked four-wheel high-range drive mode, the neutral mode, or the locked four-wheel low-range drive mode, sector plate **212** is controllably rotated to a predefined sector position associated with each mode. However, when either of the full-time four-wheel high-range or low-range drive modes are selected, power transfer system **10** is operable for modulating the clutch engagement force applied to clutch pack **164** of mode clutch assembly **46** as a function of the various sensor input signals.

[0035] Mode select mechanism **60** can take the form of any mode selector device which is under the control of the

vehicle operator for generating a mode signal indicative of the specific mode selected. In one form, the mode selector device may be in an array of dash-mounted push button switches. Alternatively, the mode selector may be a manually-operable shift lever sequentially moveable between a plurality of positions corresponding to the available operational modes which, in conjunction with a suitable electrical switch arrangement, generates a mode signal indicating the selected mode. In either form, mode select mechanism **60** offers the vehicle operator the option of deliberately choosing between the various operative drive modes.

[0036] When the locked full-time four-wheel high-range drive mode is selected, sector plate **212** is rotated to a sector position causing range sleeve **110** to move to its H position, ring gear **68** to move to its A position, and mode fork **184** to move to a position whereat pressure plate **174** is in its fully extended position. As such, the maximum clutch engagement force is exerted on clutch pack **164** and mode clutch assembly **46** is considered to be operating in a fully actuated (locked-up) condition. Thus, speed differentiation between rear output shaft **40** and front output shaft **30** is prevented. Power transfer system **10** may also include a brake which is an electrically controlled device. The brake is engaged once sector plate **212** is rotated to its sector position corresponding to the locked full-time four-wheel high-mode for locking sector plate **212** against further rotation.

[0037] If mode select mechanism **60** thereafter signals selection of the full-time four-wheel high-range drive mode, gearmotor **206** is actuated for initially rotating sector plate **212** to a position causing mode fork **184** to move to a position whereat pressure plate **174** is in its fully retracted position while range sleeve **110** is maintained in its H position and ring gear **68** is maintained in its A position. As such, the minimum clutch engagement force is exerted on clutch pack **164** such that mode clutch assembly **46** is considered to be in a non-actuated condition, thereby permitting unrestricted speed differentiation between the output shafts. However, in the full-time four-wheel high-range drive mode, mode clutch assembly **46** provides adaptive control of speed differentiation and torque biasing. Specifically, the actuated state of gearmotor **206** is continuously monitored and modulated in accordance with specific predefined relationships based on the current value of the sensor input signals. As is apparent, the magnitude of the clutch engagement force is varied by bi-directional rotation of sector plate **212** between its full-time and locked high-range sector positions.

[0038] Power transfer system **10** also permits transfer case **20** to be shifted into the neutral mode upon mode selection mechanism **60** signaling selection thereof. Controller **58** commands gearmotor **206** to rotate sector plate to a neutral sector position. In this sector position, the contour of first range slot **220** has caused range sleeve **110** to move to its N position while the contour of second range slot **226** has caused ring gear **68** to move to, or remain in, its A position. Likewise, mode slot **214** has caused mode fork **184** to move to a position whereat mode clutch assembly **46** is non-actuated.

[0039] If a full-time four-wheel low-range drive mode is made available, its selection would cause gearmotor **206** to rotate sector plate **212** to a corresponding sector position

whereat range sleeve **110** is in its L position, ring gear **68** is in its B position, and mode clutch assembly is non-actuated. Again, the contour of the range and mode slots control the coordinated movement of range forks **124**, **128** and mode fork **184** to establish the desired mode. Preferably, automatic clutch control in the full-time low-range drive mode is similar to that described the full-time four-wheel high-range drive mode. To accomplish this adaptive clutch control, sector plate **212** must be moveable from its full-time low-range sector position to a locked four-wheel low-range drive mode sector position where a maximum engagement force is applied to clutch pack **164**. As before, such rotation of sector plate **212** occurs while range sleeve **110** is maintained in its L position and ring gear **68** is maintained in its B position. Automatic control of mode clutch assembly **46** is then accomplished in the full-time four-wheel low-range drive mode to bias torque and limit slip automatically. Finally, selection of the locked four-wheel low-range drive mode signals controller **58** to rotate sector plate **212** to its corresponding sector position. In this sector position, range sleeve **110** is in its L position, ring gear **68** is in its B position, and mode fork **184** is in the position where pressure plate **174** is in its fully extended position such that mode clutch assembly **46** is locked-up. As before, the brake can be applied to hold sector plate **212** in this position so as to allow gearmotor **206** to be turned-off, thereby decreasing its on-time service.

[0040] Referring now to FIGS. 5 and 6, a modified version of transfer case **20** is designated by reference numeral **20A** and the same numbers are used to identify common components. Basically, transfer case **20A** is substantially similar in structure and function to that of transfer case **20** except that first planetary gearset **48** was arranged to provide a ratio of about 2.6 to 1 for its low-range while first planetary gearset **48A** of transfer case **20A** is adapted to provide a ratio of about 3.9 to 1 for its low-range. Specifically, in first planetary gearset **48**, first sun gear **70** has 55 teeth while ring gear **68** has 89 teeth and first planet gears **72** each have 17 teeth to define the 2.6:1 ratio. In contrast, first planetary gearset **48A** has a first sun gear **70A** with 31 teeth while ring gear **68A** has 89 teeth and first planet gears **72A** each have 29 teeth. To accommodate the different size requirements for first sun gear **70A** and first planet gears **72A**, transfer case **20A** includes a modified coupling ring **94A** which is still functional to couple sun gear **70A** for axial movement with ring gear **68A** while permitting relative rotation therebetween.

[0041] First sun gear **70A** is shown to be fixed via a splined connection **84A** for axial movement relative to quill shaft **86A**. Coupling ring **94A** is fixed to plate segment **90A** of ring gear **68A** and includes a tubular segment **236** and a radial ring segment **238**. Tubular segment **236** of coupling ring **94A** is fixed via a splined connection **240** to clutch hub **108** such that ring gear **68A** is coupled for rotation with clutch hub **108** while axially moveable relative thereto. Ring segment **238** of coupling ring **94A** is shown retained in a circumferential groove **242** formed in first sun gear **70A**. Thus, sun gear **70A** is again coupled for axial movement with ring gear **68A** between the A and B positions while still capable of rotation relative to ring gear **68A**. As with transfer case **20**, actuator assembly **54** is again provided for controlling coordinated movement of range sleeve **110** between its three distinct positions and movement of ring gear **68A**

between its two distinct positions to establish the high-range and low-range drive connections between input shaft 62 and carrier assembly 52.

[0042] Referring now to FIG. 7, a transfer case 20B is schematically shown to be a modified version of transfer case 20 such that common reference numerals are again used to identify similar components. Basically, transfer case 20B has a modified synchronized range shift mechanism 260 that combines movement of range sleeve 110 and ring gear 68 of transfer case 20 so as to provide reduced complexity and minimized packaging requirements. Specifically, planetary gear assembly 42 includes a first gearset 48B interconnected to second gearset 50 via carrier assembly 52. First gearset 48B includes a first sun gear 70B, a ring gear 68B and first planet gears 72 rotatably supported between carrier rings 78B and 80. Sun gear 70B is shown to have a coupling ring 262 fixed thereto which includes an inner cylindrical rim segment 264 and an outer cylindrical rim segment 266 interconnected by a plate segment 268. Inner rim segment 264 has a set of first internal spline teeth 270 which are axially offset from from a set of second internal spline teeth 272. Also, input shaft 62B is shown to include a first set of external spline teeth 274 and a second set of external spline teeth 276. A radial lug 278 extending outwardly from outer ring segment 266 is retained in a circumferential groove 280 formed in range sleeve 110B. Ring gear 68B is shown to be fixed to one end of range sleeve 110B such that it and sun gear 70B are axially moveable with range sleeve 110B.

[0043] Range sleeve 110B is axially moveable between three distinct range positions (L, N, H) via movement of range fork 124 upon controlled actuation of actuator assembly 54. Range sleeve 110B includes internal clutch teeth 282 which are in constant mesh with external teeth 284 formed on a low hub 286. Low hub 286 is rotatably supported on a low clutch plate 288 that is fixed to housing 66. A low synchronizer assembly 290 is disposed between low hub 286 and low clutch plate 288 and functions to establish speed synchronization therebetween prior to permitting clutch teeth 282 of range sleeve 110B to enter into engagement with clutch teeth 292 on low clutch plate 288 during movement of range sleeve 110B into its L position. When a four-wheel low-range drive mode is selected, gearmotor 206 rotates sector plate 212 of drive mechanism 208 for causing range fork 124 to move range sleeve 110B to its L position. Such movement of range sleeve 110B causes both sets of clutch teeth 270 and 272 on coupling ring 262 to meshingly engage corresponding sets of clutch teeth 274 and 276 on input shaft 62B while also causes its clutch teeth 282 to engage clutch teeth 292 on low clutch plate 288. Thus, sun gear 70B is driven by input shaft 62B and ring gear 68B is braked by housing 66 against rotation such that carrier assembly 52 is driven at a reduced speed ratio. First planetary gearset 48B can be arranged to provide any suitable reduction ratio including, without limitation, either of the 2.6:1 or 3.9:1 ratio previously disclosed.

[0044] With continued reference to FIG. 7, synchronized range shift mechanism 260 is further shown to include a high clutch hub 300 that is rotatably supported on coupling ring 262 and which has external spline teeth 302 in constant mesh with internal clutch teeth 304 formed in outer ring segment 266 of coupling ring 262. First carrier ring 78B is shown to include clutch teeth 306 that are aligned to receive clutch teeth 304 of coupling ring 262 upon movement of range

sleeve 110B to its H position. A high synchronizer assembly 308 is disposed between hub 300 and carrier ring 78B and functions to establish speed synchronization between carrier assembly 52 and sun gear 70B prior to engagement of coupling ring teeth 304 with carrier ring teeth 306. When it is desired to establish a four-wheel high-range drive mode, range sleeve 110B is moved to its H position where teeth 270 on coupling ring 262 engage teeth 276 on input shaft 62B such that sun gear 70B is driven by input shaft 62B. Also, upon synchronization, clutch teeth 304 on coupling ring 262 engages clutch teeth 306 on first carrier ring 78B such that a direct drive connection between input shaft 62B and carrier assembly 52 is established. Range sleeve 110B is shown in its N position with coupling ring 262 disengaged from input shaft 62B.

[0045] The foregoing discussion discloses and describes the preferred embodiments for the present invention. However, one skilled in the art will readily recognize from such discussion, and from the accompanying drawings and claims, various changes, modifications and variations can be made therein without departing from the true spirit and fair scope of the invention as defined in the following claims.

What is claimed is:

1. A transfer case comprising:

an input shaft;

first and second output shafts;

a planetary gear assembly interconnecting said input shaft to said first and second output shafts and including first and second gearsets having a common carrier assembly, said first gearset including a first sun gear driven by said input shaft, a ring gear, and a first planet gear supported by said carrier assembly and meshed with said first sun gear and said ring gear, said second gearset including a second sun gear connected to said first output shaft, a third sun gear operably connected to said second output shaft, a second planet gear supported by said carrier assembly and meshed with said second sun gear, and a third planet gear supported by said carrier assembly and meshed with said third sun gear and said second planet gear;

a range sleeve fixed to said ring gear for movement between a high-range position and a low-range position, said first sun gear is retained for movement with said range sleeve while permitting relative rotation between said ring gear and said first sun gear, said range sleeve is operable in its high-range position to couple said first sun gear to carrier assembly and is further operable in its low-range position to couple said ring gear to a stationary member and release said first sun gear for rotation relative to said carrier assembly; and

a shift mechanism for moving said range sleeve between its high-range and low-range positions.

2. The transfer case of claim 1 wherein said first sun gear is fixed to a coupling ring that is splined for rotation with and axial movement relative to said input shaft in response to movement of said range sleeve.

3. The transfer case of claim 2 wherein said coupling ring includes a set of clutch teeth that are releaseably engageable

with clutch teeth formed on a carrier ring of said carrier assembly when said range sleeve is in its high-range position.

4. The transfer case of claim 3 wherein said range clutch includes a clutch hub that is rotatably supported on said coupling ring and has clutch teeth in constant meshed engagement with said clutch teeth of said coupling ring, and further comprising a synchronizer that is operably disposed between said clutch hub and said carrier ring.

5. The transfer case of claim 1 wherein said range clutch includes a clutch plate fixed to said stationary member having clutch teeth adapted for engagement with clutch teeth on said range sleeve when said range sleeve is in its low-range position.

6. The transfer case of claim 5 further comprising a synchronizer operably disposed between said clutch plate and said range sleeve.

7. The transfer case of claim 6 wherein said range sleeve is further operable in a neutral position whereat said coupling ring is uncoupled from said input shaft, said ring gear is uncoupled from said stationary member, and said first sun gear is uncoupled from said carrier assembly.

8. A transfer case comprising:

an input shaft;

first and second output shafts;

a first planetary gearset including a first sun gear, a ring gear, and a first planet gear rotatably supported by a carrier and meshed with said first sun gear and said ring gear;

a second planetary gearset including a second sun gear driving said first output shaft, a third sun gear driving said second output shaft, a second planet gear rotatably supported by said carrier and meshed with said second sun gear, and a third planet gear rotatably supported by said carrier and meshed with said second sun gear and said second planet gear;

a first coupling fixed for rotation with said first sun gear;

a second coupling fixed for rotation with said ring gear, said second coupling moveable between a first position and a second position;

a third coupling interconnecting said first coupling member for common movement with said second coupling member while permitting relative rotation therebetween; and

a shift mechanism for moving said second coupling between its first and second positions, said second coupling operable in its first position to couple said first coupling to said input shaft and said third coupling to

said carrier, said second coupling is further operable in its second position to couple said first coupling to said input shaft and couple said ring gear to a non-rotary member.

9. The transfer case of claim 8 wherein said second coupling is further moveable to a third position whereat said first coupling is uncoupled from said input shaft and said third coupling is uncoupled from said carrier and said second coupling is uncoupled from said non-rotary member.

10. The transfer case of claim 8 wherein said input shaft has first and second splines and said first coupling has first and second splines, wherein movement of said second coupling to its first position causes corresponding movement of said first coupling such that its first splines engage said second splines on said input shaft, and wherein movement of said second coupling to its second positions causes said first coupling to move to a position whereat its second splines engage said first splines on said input shaft.

11. The transfer case of claim 8 wherein said third coupling includes clutch teeth that are releaseably engageable with clutch teeth formed on said carrier when said second coupling is in its first position.

12. The transfer case of claim 11 further comprising a synchronizer operably disposed between said clutch teeth of said third coupling and said clutch teeth on said carrier.

13. The transfer case of claim 8 wherein said second coupling has clutch teeth operable to move into meshed engagement with clutch teeth on said non-rotary member when said second coupling is located in its second position.

14. The transfer case of claim 13 further comprising a synchronizer operably disposed between said clutch teeth on said second coupling and said clutch teeth on said non-rotary member.

15. The transfer case of claim 8 further comprising a mode clutch for controlling speed differentiation and torque biasing between said first and second output shafts.

16. The transfer case of claim 15 wherein said mode clutch includes a friction clutch assembly operably disposed between said first and second output shafts, and a power-operated clutch actuator for adaptively controlling the magnitude of a clutch engagement force exerted on said friction clutch assembly.

17. The transfer case of claim 16 wherein said power-operated clutch actuator includes a cam member, an electric motor for moving said cam member, a thrust mechanism for applying said clutch engagement force in response to movement of said cam member, said electric motor receiving control signals from a controller based on sensor signals from at least one sensor measuring a particular operating characteristic.

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